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TRANSLATION

ELIMINATING AIRCRAFT ENGINE VIBRATION

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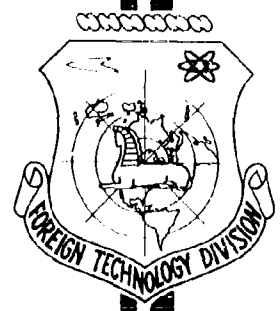
M. Ye. Levit and Yu. A. Kolosov

FOREIGN TECHNOLOGY DIVISION

AIR FORCE SYSTEMS COMMAND

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ELIMINATING AIRCRAFT ENGINE VIBRATION

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With the development of the new technology, a great deal of attention is being directed to the study of elastic oscillations — their calculation and prevention. The works of many researchers and plant engineers are devoted to this problem.

In spite of the decided achievements in this area, the study of the causes of machine vibration remains a complicated problem, because it is difficult to take into account and evaluate the influence of many and varied factors. Among these are various types of deformation, asymmetric rigidity, yielding of supports, etc. However, one effect is certain and basic; it is imbalance in high-rpm assemblies, which leads the system to vibrate with an amplitude which is a function of the degree of imbalance.

The vibrations transmitted to the machine can cause resonance in its bracing units, disrupt the normal operation of fittings and instruments, and considerably lower its operating resources. The oscillations have a substantial effect on the maintenance personnel. For example, oscillations above 50 cps can be propagated through the human body

without any noticeable fading. This heightens their negative effect on the central nervous system and on the organism as a whole. Under the action of vibrations, serious disorders result in the human organism.

Inasmuch as machine vibrations are first of all a function of rotor imbalance, the development of methods for higher-precision balancing and the creation of the necessary arrangements to guarantee vibrationless operation take on great importance.

Vibration in gas-turbine engines and ways of reducing it

The systematic improvement of the design of gas-turbine engines introduced many methods of combatting vibrations and, owing to the design features of the engines (one piece and multipiece rotors), oscillations are observed which have varied character and form, however, the methods of eliminating them are basically general. They come down to determining the magnitude of the vibration overload of the engine and rebalancing of the rotors. Test stands are used, and each experiment includes complete dismantling of the engine. This method, although it corresponds to actual operating conditions and takes into account dynamic rotor deflections, temperature and axial force, it has the essential disadvantage of requiring overhauling of the engine, which leads to a change in the initial conditions (the precision of parts, clearances in joints, bearing seatings, etc.).

At the present time, the method of reducing the amplitude of vibrations which exceed the limits of engineering conditions is chiefly the following:

1. Damping of the oscillations of the system and tuning them out.

of the resonance conditions of operation.

2. Increasing the quality and accuracy of dynamic equilibrium, allowing the vibrations of the system to be decreased.

Without touching on the first of these, let us say that so far the basis of the method of dynamic balancing of rotating bodies is the assumption of their absolute rigidity. Therefore, any number of unbalanced centrifugal forces is reduced to two forces, lying usually at the outer planes. Because of this, the balancing machines are fitted with two supports, on which the body to be balanced turns at low rpm (400 to 800 rpm). By choosing the magnitude and location for the balancing weight, the action of the unbalanced centrifugal force is eliminated for one support. The same operation is carried out for the other support. However, the use of high rpm machines shows that even comparatively rigid rotors are deformed during transition to higher speeds. Deflections increase sharply and the rotor takes on the so-called elastic imbalance. This imbalance leads to a strong increase in the amplitude of oscillations. Therefore, a method for balancing high rpm elastic systems having great mass is a real problem.

If rotors with great deformation belong to the category of elastic systems, then they must be balanced at operating rpm. However, this kind of balancing is very difficult under the present conditions. It requires high power (25,000 to 50,000 kw), complicated apparatus and a comparatively long time for balancing.

In this connection the authors, with the participation of engineers I. I. Bayenko and G. K. Devyatov, have studied a new method of balancing full-scale rotors at working rpm with a small expenditure of power for their running. This method differs from the existing

ones in that the correcting weights are introduced not only at the outer planes, but also at one or more of the driving planes along the length of the rotor. With this, rotor deformation (deflection of elastic line) is reduced. The location and magnitude of imbalance is determined by special pick-ups at the operating speed of the rotor.

Although it is impossible to eliminate vibration completely in turbo-jet engines, the proposed method ensures that it will be lowered to an acceptable level.

The method of balancing and the equipment used

The balancing of full-scale engines and their assemblies by the proposed method is carried out on a special stand, which allows balancing to be done at operating rpm. The rotor is set up together with the housing, i.e., on its own supports.

The schematic diagram of the stand (Fig. 1) provides for the location of the object of study in the chamber. Here the suspension conditions for the object are close to the actual ones, in order to preserve its frequency characteristic insofar as possible.

The rotor is started up by a d-c electric motor through a gear box. In order to exclude the effect of driving on the oscillations of the object, there is a free-running clutch in the system.

The chamber in which the object of study is installed can be of varied design. A vacuum can be created in it (1 to 3 mm Hg), which allows the power required to turn the huge rotor to be much lower. For a number of objects, a separate vacuum chamber need not be made, but having provided the necessary seals, a vacuum is created in the engine housing. For example, when examining vibrations in the VK-1 engine on an MAI-1 stand, the compressor housing, fitted with three

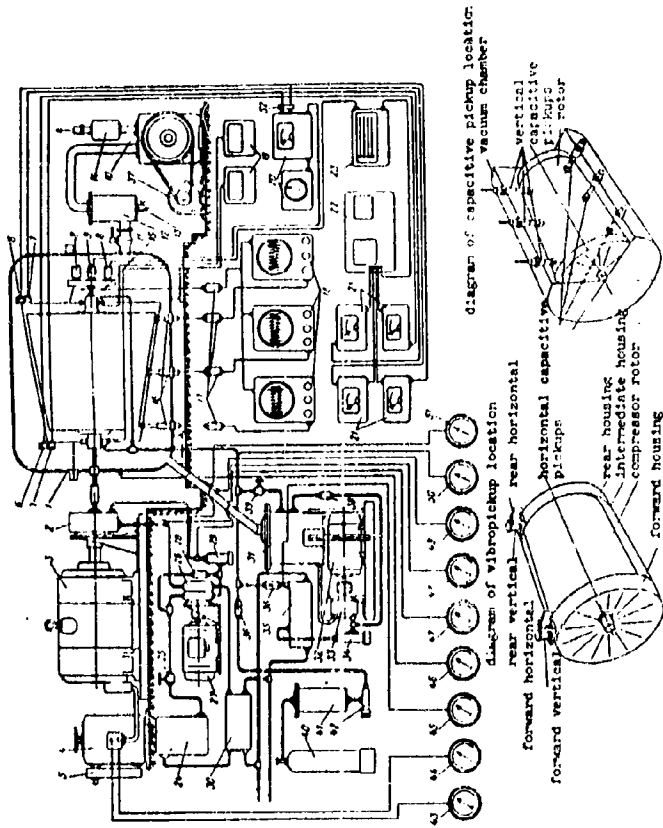


Fig. 1. Schematic diagram of vacuum stand for vibration measurement and balancing.
 1) vacuum chamber; 2) gear box; 3) d-c electric motor; 4) starter; 5) shunt rheostat;
 6) vertical vibropickup; 7) horizontal vibropickup; 8) tachometer pickup; 9) phase-
 meter generator; 10) vacuum pump; 11) vacuum valve; 12) trap; 13) overflow cock; 14)
 oil collector; 15) capacitive pickup of place marker; 16) capacitive pickup of stage;
 17) preamplifier; 18) oscillograph; 19) bearing-temperature meter; 20) phase meter;
 21) vibration meter; 22) loop oscillograph; 23) rotor tachometer; 24) oil tank;
 25) stopcock; 26) oil pump; 27) electric motor; 28) pressure regulator; 29) filter;
 30) radiator; 31) oil tank; 32) electric motor; 33) pump; 34) filter; 35) radiator;
 36) reverse channel; 37) electric motor; 38) pressure regulator; 39) bypass cock;
 40) compressed air tank; 41) oil tank; 42) filter; 43) ammeter; 44) volt meter;
 45) pressure gage; 46) vacuum meter; 47) pressure gage; 48-51) thermometers; 52) switch.

sealing discs, a cup-type seal along the compressor shaft and plugs, served successfully as a vacuum chamber (Fig. 2). But sometimes the housing cannot be sealed reliably enough. It is better to use a special vacuum chamber then.

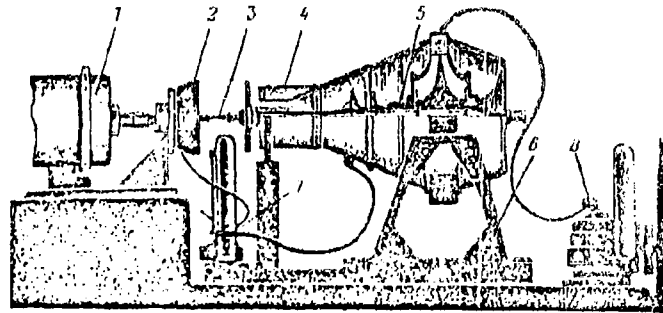


Fig. 2. Diagram of vacuum apparatus with a VK-1 engine. 1) electric motor; 2) gear; 3) free-running clutch; 4) engine to be studied; 5) sealing ring; 6) frame; 7) lubrication system; 8) vacuum system.

An MAI-3 stand was built in order to study the vibrations of the axial compressors, where the vacuum chamber was of self-contained design.

It is a steel cylinder closed at the ends by discs (Fig. 3). There is a cup seal in the forward disc which makes it airtight along the drive shaft. In addition, there is a hatch in the disc for convenience in mounting the balancing weights. The disc is bolted to the flange of the chamber. This connection is made airtight with a vacuum-rubber gasket 2 to 3 mm in diameter.

A special nacelle is mounted on the rear disc, through which all electrical contacts and oil lines exit, and to which the pipes from the vacuum pumps are connected. In this same disc there is also an opening for the main drain of the oil system.

Along the outer surface of the chamber are several strong

stringers connected by corner plates. The stringers are reinforced, four adjustable journals are mounted on them, by which the chamber is secured to the frame, which is mounted on the base plate. In addition to this, there is in the chamber a hatch for ease in getting to the rotor and several tens of openings for mounting the capacitive pickups. Two annular frames are welded inside the chamber for mounting the object to be studied.

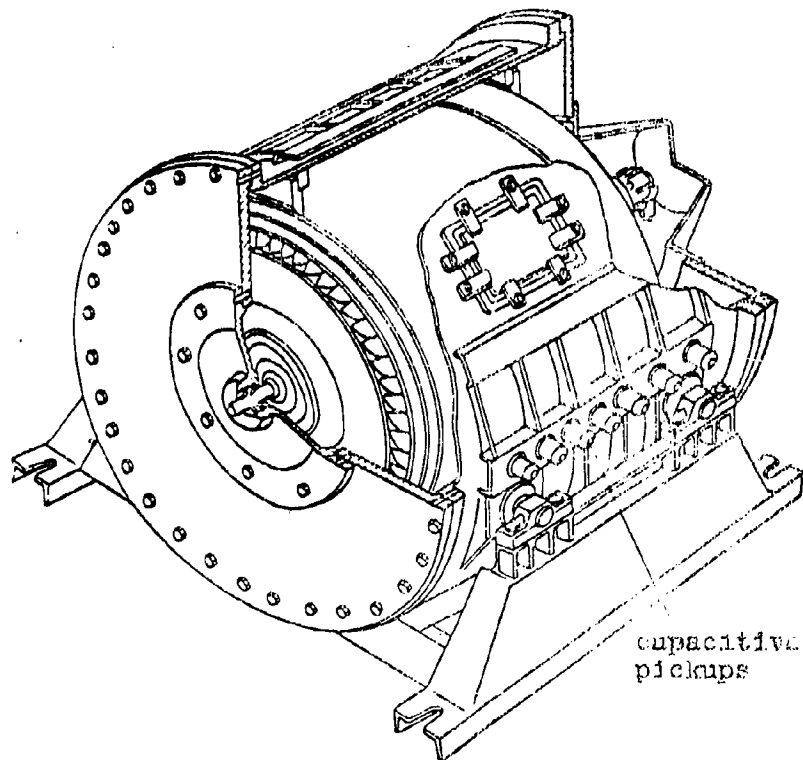


Fig. 3. Over-all view of vacuum chamber with axial compressor.

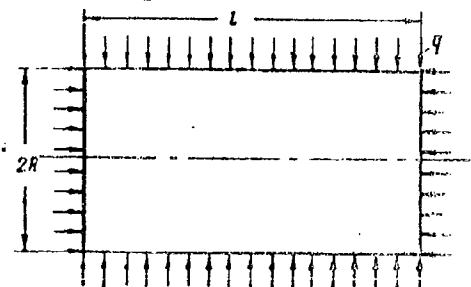


Fig. 4. Calculated stress diagram of the vacuum chamber.

The chamber of the stand, in the calculation of both rigidity and stability, is considered an axially symmetric shell subjected to the simultaneous action of axial compression and uniform transverse pressure (Fig. 4). Under the action of these forces, the shell can retain its cylindrical shape only up to known (practical) load values. At critical loads this equilibrium shape becomes unstable and the shell can bulge.

Let us assume that the shell is thin and its edges are free of support. For this case S. P. Timoshenko [1] puts forth a simplified formula for determining the critical load

$$q_{cr} = \frac{Eb}{R} \frac{1}{h^2 + 0.5 \left(\frac{\pi R}{l}\right)^2} \left\{ \frac{1}{\left[h^2 \left(\frac{l}{\pi R}\right)^2 + 1 \right]^2} + \frac{h^2}{12R^2(1-\mu^2)} \left[h^2 + \left(\frac{\pi R}{l}\right)^2 \right]^2 \right\}.$$

where $E = 2.1 \cdot 10^6$ kg/cm² is Young's modulus for the material of the chamber; δ the thickness of the shell in cm; R the radius of the shell (60 cm); l the length of the shell in cm; μ Poisson's ratio; q atmospheric pressure in kg/cm²; and h the number of parts about the periphery at which the shell bulges. It is determined as a function of the ratios $\delta/2R$ and $l/2R$ according to the graph in Fig. 251 of Timoshenko's work [1].

In our case a high vacuum is created inside the chamber with atmospheric pressure on the outside. Therefore, the maximum pressure drop q will equal 1 kg/cm².

In order to increase the stability reserve, longitudinal profile ribs (stringers) and transverse ribs (frames) are welded to the shell.

When calculating stability, the presence of the reinforcements should be taken into account. However, calculation of the stability of a reinforced shell is very complicated and, therefore, we shall limit ourselves to the calculation of stability of an unreinforced

shell, taking into account its length and thickness.

Calculations have been made for various values of δ and l . The results of the calculations show that when $l = 150$ cm and $\delta = 1$ cm even an unreinforced shell has a stability reserve

$$\frac{q_{cr}}{q} = 29.$$

The introduction of reinforcing ribs increases the stability reserve even more.

The drive system is self-contained owing to which it is possible to turn various rotors without modifying it.

Turning of the object to be examined is accomplished by a 130-kw d-c electric motor operating at 1000 rpm. Smoothness of turning is attained by the starting and shunt rheostats. The shaft of the electric motor is connected to the gear box by a slit spring; the gear box is mounted on a special stand on a single base plate with the electric motor.

The drive shaft of the gear box is connected to the shaft of the free-running clutch, which in this case is mounted in the rear nozzle of the compressor. The free-running clutch is necessary to eliminate the effect of driving when studying the causes of vibration.

The drive shaft of the free-running clutch passes through a cup packing. Taking into account that the power required to turn the rotors of modern turbo-jet engines on MAI vacuum vibromeasuring stands is from 50 to 60 kw, shafts of up to 18 mm in diameter should be used, which allows the life of the cups to be prolonged.

The shaft driving the object under study operates by twisting. The most dangerous time is when the spring is wound up in starting the rotor.

The equation of motion of the apparatus for the case of a constant reduced moment of inertia of the rotor will have the form

$$M_d - M_c = J_M \frac{d\omega}{dt},$$

where M_d is the reduced moment of driving forces in kgcm; M_c the reduced drag moment in kgcm, J_M the polar moment of inertia of the rotor, equal to 220 kgcm², and $\frac{d\omega}{dt}$ the angular acceleration in 1/sec².

Since the rotor is in a vacuum the drag moment (from the friction of the rotor against the air) can be ignored. This is all the more valid at the moment of starting the rotor, when the peripheral velocities of the given rotor are not yet high. Therefore,

$$M_d = J_M \frac{d\omega}{dt} = M_{cr}$$

The winding tension in the spring

$$\sigma = \frac{M_{cr}}{W_{cr}} \approx R_{add},$$

where $W_{cr} \approx 0.2d^3$.

From the operating conditions we take $R_{add} = 1500$ kg/cm² for alloyed steel with $\sigma_b = 10,000$ kg/cm². Hence we find

$$\sigma = \sqrt{\frac{M_{cr}}{0.2R_{add}}} = \sqrt{\frac{I_M \frac{d\omega}{dt}}{300}}$$

Let us assume that the rotor must be spun up to $n_{max} = 8000$ rpm, or $\omega_{max} = 840$ /sec.

For the case when the starting time $t = 60$ sec, we obtain

$$\begin{aligned} \text{a) } \tan \alpha'_{av} &= \left(\frac{d\omega}{dt} \right)_{1/av} = 14/\text{sec}^2; \quad M_{cr} = J_M \frac{d\omega}{dt} = 220 \cdot 14 = \\ &= 3040 \text{ kgcm} \end{aligned}$$

$$d = \sqrt[3]{\frac{3030}{300}} = 2.2 \text{ cm} = 22 \text{ mm};$$

$$b) \tan \alpha_{\max}^I = \left(\frac{d\omega}{dt}\right)_{\max} = 24 \text{ 1/360}^{\text{sec}}; M_{\text{cr}} = 220 \cdot 24 = 5280 \text{ kgcm}$$

$$d = \sqrt[3]{\frac{5280}{300}} = 2.6 \text{ cm} = 26 \text{ mm}.$$

For the case when the starting time $t = 2$ min, we obtain

$$a) \left(\frac{d\omega}{dt}\right)_{2 \text{ av}} = \tan \alpha_{\text{av}}^{\text{II}} = 7/\text{sec}^2; M_{\text{cr}} = 220 \cdot 7 = 1540 \text{ kgcm}$$

$$d = \sqrt[3]{\frac{1540}{300}} = 1.73 \approx 18 \text{ mm};$$

$$b) \left(\frac{d\omega}{dt}\right)_{2 \text{ max}} = \tan \alpha_{\max}^{\text{II}} = 12/\text{sec}^2; M_{\text{cr}} = 220 \cdot 12 = 2640 \text{ kgcm}$$

$$d = \sqrt[3]{\frac{2640}{300}} = 2.1 \text{ cm} = 21 \text{ mm}.$$

Considering that the starting time can be increased to $t > 2$ min, the spring diameter can be limited to 18 mm.

The power expended in turning the rotor is a function of the design and dimensions of the rotor, and also of the depth of the vacuum in its housing. The main part of this power is expended in overcoming friction of the rotor against the air. The friction power N for a wheel with a given relationship of geometric dimensions is a function of the angular velocity ω , the outer diameter of the wheel D , and the density of the air ρ .

$$N = f(\omega, D, \rho).$$

The approximate calculations of the power and the results of the experiments confirming them are shown in the graph in Fig. 5 for turbo-jet engines with centrifugal compressors. In order to wind up the rotor of an axial compressor of a powerful turbo-jet engine to operating rpm when the vacuum in the chamber equals 1 to 2 mm Hg, about 60 kw of power is required.

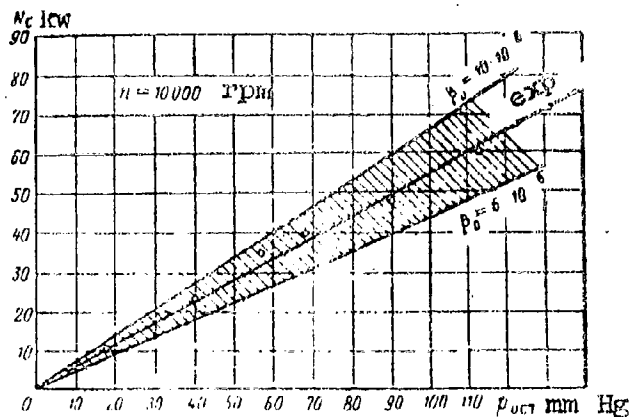


Fig. 5. Power required to turn rotor versus chamber vacuum.

Calculation of power is made difficult by the fact that the disc-loss factors in a vacuum on the order of 1 to 3 mm Hg are obtained experimentally.

The oil system consists of two autonomous circulating oil systems (Fig. 1). The first is for lubrication of the bearings in the vacuum zone. The first is called the gear-box oil system, and the second the vacuum oil system.

In the gear-box oil system there is a two-stage oil pump 26; the oil comes from the tank 24 through the stopcock 25. The pump is driven by an electric motor 3. The oil is fed to the bearings from the pressure stage of the pump through the filter 29. A pressure regulator 28 serves to control the working pressure of the service mechanisms. From the output of the pump-out stage the oil is fed through a radiator 30 to the service tank 24. The special feature of the vacuum oil system is the task it performs: the lubrication and cooling of bearings in a vacuum on the order of 1 to 2 mm Hg.

Much work was done in choosing the oil-system plan and the kind of lubricant which would work at this pressure. As a result of the

investigations several efficient plans were chosen (one of which is shown here) and only one type of lubricant-silicone (fluid Nos. 5 and 6).

A single-stage pump 33 is installed in the vacuum oil system to supply lubrication to the rotor bearings. The pump is driven by an electric motor. From the service tank 31 the lubricant is fed into the pump and then, passing through the filter 34 and the radiator 35, it is fed to lubricate the bearings.

The working pressure in the system is controlled by a pressure regulator 38 and a bypass cock 39. The lubricant is returned to the service tank by a pipe.

In the case of forced stopping of the vacuum oil system there is an emergency device which works as follows: compressed air, directed from a cylinder 40 into a tank 41, forces the lubricant there through the filter 42 and the reserve channel 36 to the bearings.

The vacuum system is designed to create and maintain the required evacuation in the vacuum chamber. This is achieved by rational choice in the design of the working space (vacuum chamber), by having simple and reliable seals in all joints and hatches of the chamber and in all vacuum lines and, finally, by using the proper type of vacuum pump. In addition to this, it is necessary to install in the working space as few as possible units, parts and lines of materials which vaporize easily in a vacuum (rubber, vinyl chloride, plastolite, ebonite, wires in all types of insulation, etc.). This is especially important if the vacuum pump does not have sufficient reserve output, because an unavoidable inflow of air through the seal into the working space and a large quantity of easily vaporized material can interfere with the creation of the required vacuum.

The measuring apparatus of the stand is divided into two independent groups.

The first group contains the apparatus for measuring and recording the vibrations: type MV-21 vibropickups, SIV-3 vibration meters and an MPO-2 oscillograph. This is IP-1 or IP-2 apparatus, for recording the change in magnitude and direction of rotor deflection. This equipment was developed and approved by the Moscow Aviation Institute (MAI).

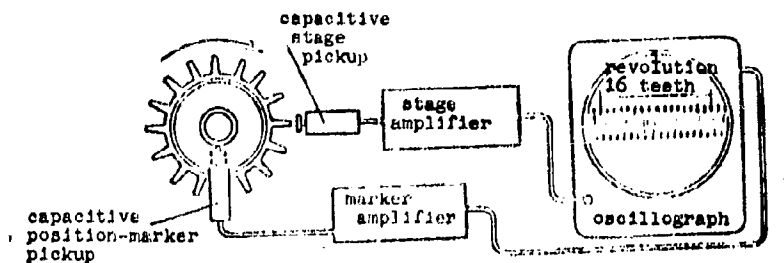


Fig. 6. Block diagram of IP-1 apparatus.

The second group contains the entire measuring apparatus, which allows the operation of all auxiliary systems of the stand to be controlled: the vacuum, drive, oil systems, etc. The instruments in this group are standard vacuum meters, ammeters, volt meters, pressure gages, and thermometers, so it is not necessary to describe them in detail.

Measuring and recording of vibrations is accomplished on equipment used by the aircraft industry; the testing and calibration of this apparatus is also done by generally accepted methods. Therefore, we shall only describe the operation and purpose of the subgroup of measuring equipment used for recording the change in magnitude and direction of rotor deflection, i.e., the basic apparatus in rotor balancing or studying the causes of increased machine vibration.

As was mentioned above, there are two types of this apparatus: IP-1 and IP-2. The type IP-1 measuring apparatus (Fig. 6) consists of a capacitive stage pickup, a capacitive position-marker pickup, two amplifiers and an oscillograph. As is apparent from Fig. 7, the IP-1 unit is very simple to produce and make it possible to obtain a sufficiently clear qualitative picture of rotor deflection and the change in its direction over the entire range of operating revolutions (Fig. 8). The disadvantage of this apparatus is the complexity of its calibration, when it is necessary to make a quantitative determination of the magnitude of deflection (see IP-1 calibration). Now let us examine how the magnitude and direction of rotor deflection are determined by using the IP-1.

The capacitive stage pickups are installed in several rotation zones of the rotor under investigation, usually opposite the stages; hence their name. The sensing elements of the pickups can be moved toward the rotor axis with accuracy up to 0,01 mm. This is necessary for calibrating the apparatus. The working gap between the blades of the stage and sensing element can be set up to 5 mm, depending upon the structure of the object. The capacitive pickups are connected through an amplifier to the oscillograph.

When the rotor turns, the blades of the stage, passing by the sensing element of the capacitive pickup, change its capacitance, and a pulse appears on the oscillograph screen. Because of the continuity of the passing of the blades by the pickup, we can see on the oscillograph screen a sine curve of the so-called carrier frequency.

In deflection, the sine curve of the carrier-frequency envelope is clearly visible on the screen. Spread in the amplitude of the envelope corresponds to double deflection of the rotor in a given

area (Fig. 8).

Thus we determine the relative magnitude of the deflection and its variations with respect to rpm, that is, the qualitative picture of the elastic line (to determine the actual magnitude of deflection see "Calibration of IP-1 Apparatus"). In order to obtain the full picture of the elastic line of the rotor, it is also necessary to know the direction of deflection relative to any previously assigned point of reference. An additional capacitive pickup, a position marker, is used for this.

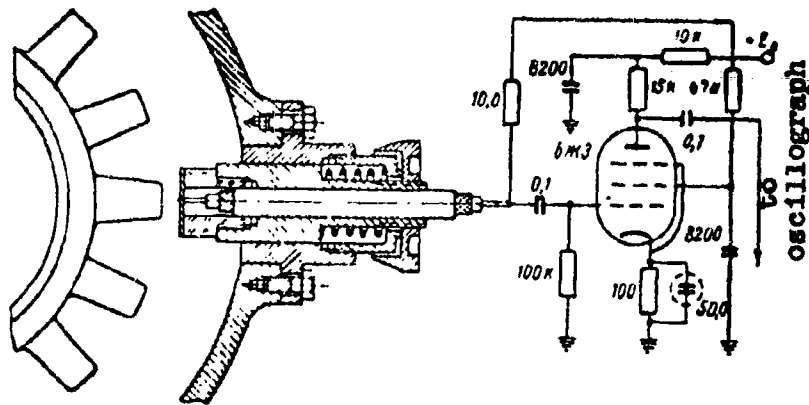
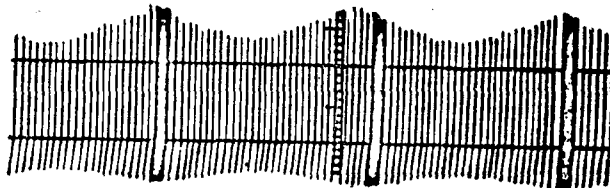


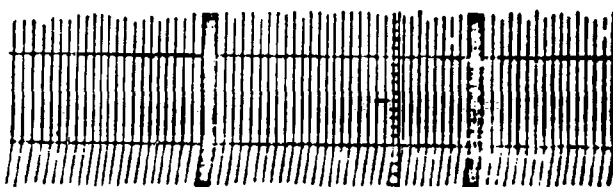
Fig. 7. Schematic of capacitive pickup and amplifier unit of type IP-1 apparatus.

The direction of deflection is determined as follows. A metal pin is mounted radially on the rotor shaft in an arbitrary position; the capacitive position-marker pickup is mounted in the plane of rotation of this pin (Fig. 6). The pickup is connected through an amplifier to the modulating electrode of the oscillograph. When the rotor turns, the pin passes by the sensing element of the capacitive pickup and one pulse per revolution is seen on the screen. It is obvious that for each revolution of the rotor we shall see a pulse which characterizes a fully determined marker position relative to the maximum of the envelope of deflection. Thus it is possible to

determine with sufficient accuracy (to one blade) the number of blades from the maximum of the envelope to the position-marker pulse, i.e., to determine with accuracy to one blade the direction of rotor deflection in a given plane.



a)



b)

Fig. 8. Photograph of oscillograph screen.
a) in the presence of deflection; b) in the absence of deflection.

In order to determine the actual magnitude of rotor deflection, the IP-1 apparatus is calibrated.

At the basis of the method lies the balancing under resonance conditions of the system in the range of operating rpm. Therefore, in rotor balancing we are interested in its maximum deflection and the shape of the elastic line. Therefore, the capacitive apparatus is calibrated for just those conditions. The calibration technique is as follows.

Having recorded on the grid of the oscillograph screen the maximum position of the envelope, we depart from resonance by 50 to 100 rpm up or down. In this the amplitude of the envelope decreases or vanishes. Departing thus from the resonance maximum, we begin to lead the capacitive pickup toward the stage. The signal from the

blades of the stage will increase. Having brought up the signal on the screen to the former maximum value of the deflection envelope, we stop moving the pickup toward the stage and measure the distance moved with a micrometer. This value will be equal to twice the deflection of the rotor in a given plane of measuring.

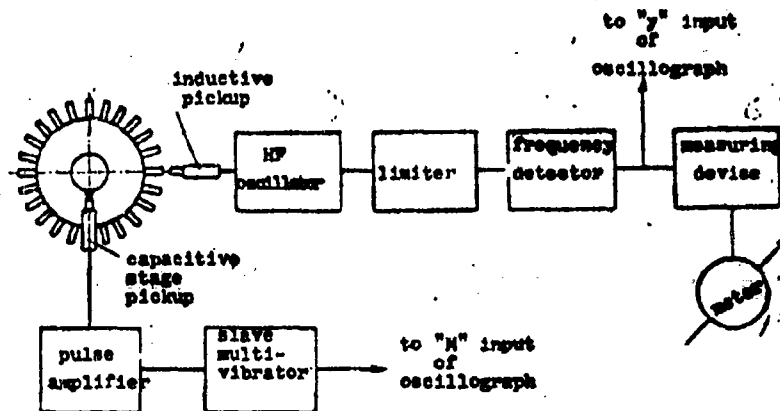


Fig. 9. Block diagram of IP-2 apparatus.

The IP-1 apparatus is used when the structure and parameters of the object to be balanced and the balancing stand allow the apparatus to be calibrated by the above method. In this case it should be noted that for balancing it is sufficient to have a qualitative picture of the deflections; their absolute value is important only when making other studies.

Thus it is sufficient to equip the stand with IP-1 apparatus for balancing at operating rpm.

The IP-2 is a further development of the apparatus for measuring the magnitude and direction of deflections. The distinguishing feature of the IP-2 is the independence of the results of measurements from the speed of the rotor from hundreds of revolutions per minute to several tens of thousands. Owing to this, calibration is considerably simplified.

Figure 9 shows a block diagram of the IP-2. A pickup made in the form of a small inductance coil is mounted at a distance of 1.5 to 2.5 mm from the ends of the rotor blades. The size of the gap is determined by the structure of the object, i.e., by the working gap between the rotor and the housing, and also by the sensitivity of the pickup.

The inductance of the pickup is part of the high-frequency oscillator circuit. When a blade passes by the pickup its inductance changes, and, consequently, the frequency of the h-f oscillator. The frequency deviation of the latter is a function of the distance between the pickup and the blade ends, i.e., in the final analysis a function of the deflection of the rotor shaft and not a function of the rotor speed.

From the high-frequency oscillator, after passing through a limiter, the frequency-modulated voltage enters the frequency detector, from the load of which the voltage, varying in amplitude in accordance with the change in frequency, enters the measuring part of the circuit and the vertical amplifier of the oscillograph.

The processes taking place in the circuit of the instrument are shown in Fig. 10, where

$$T_b = \frac{60}{nz}; T_r = \frac{60}{n};$$

n is the rotor speed in revolutions per minute; and z is the number of blades of the given rotor stage.

From Fig. 10 it is apparent that a voltage increment ΔU_M appears in the presence of deflection. The measuring part of the circuit measures this increment and thus allows the meter to register the amount of deflection.

In order to determine the location of the so-called "heavy place" with an oscillograph, bright marks are used which correspond to each revolution. To accomplish this a metal pin is mounted on the rotor shaft and a capacitive pickup is installed in its plane of rotation (see Fig. 9). When the pin passes by the pickup plate, there is a rapid change in its capacitance, which is transformed into a voltage pulse. After amplification this voltage enters a slave multivibrator. From the multivibrator the pulse enters the modulating electrode of the cathode ray tube.

Balancing at Operating RPM

The possibility of observing changes in rotor deflection in several places along its length, over the entire range of operating rpm, allows an effective method of balancing to be put forward.

The basis of this method is balancing with respect to several driving planes at resonance conditions of the system in the range of operating rpm of the rotor.

Thus we are interested in the shifting of the rotor, its deflection at resonance and near-resonance conditions, where maximum vibration takes place.

From these conditions comes the following balancing order:

1. The setting up of the rotor in the vacuum chamber together with the framework. The supports should be such that the frequency characteristic of the rotor with framework is close to its frequency characteristics on the full-scale object.

2. The connection of all necessary systems to the vacuum chamber and the checking of their operation.

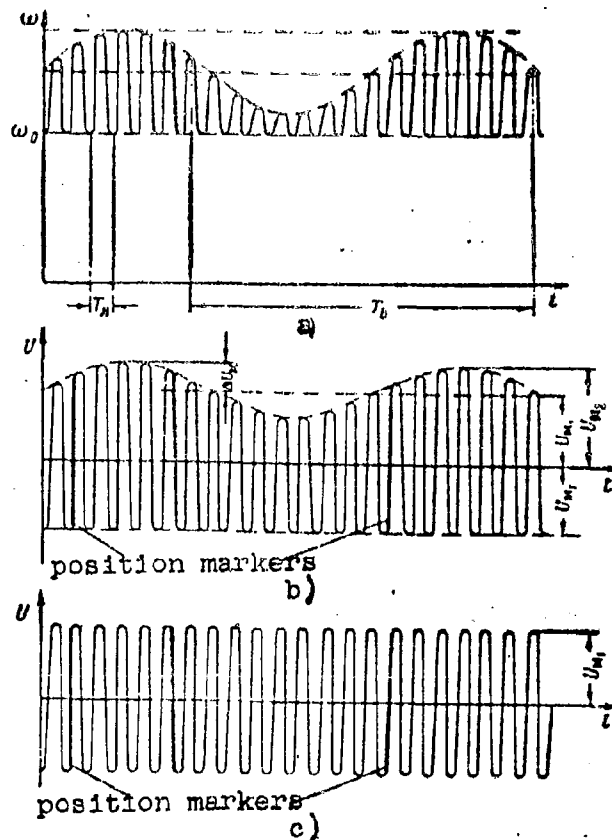


Fig. 10. Change in frequency and voltage of IP-2 unit as a function of deflection.
 a) change in frequency of h-f oscillator in the presence of deflection; b) voltage on oscillograph screen in the presence of deflection; c) voltage on oscillograph screen in the absence of deflection.

3. The switching on of the vacuum pump (the required evacuation of the chamber).
4. The switching on of the oil systems and the current to the measuring apparatus by obtaining the required vacuum in the chamber.
5. The final check of the operation of all systems.
6. The switching on of the drive motor. The starting rheostat regulates the smooth starting of the rotor. When necessary starting can be in the areas of constant rpm.

7. While gathering speed, vibrations and meter readings are observed at rated and below rated rpm; the most interesting times being recorded on film, if necessary.

8. In order to increase the reliability of the results, the start is repeated without changing the rotor load until stable results are obtained. This depends upon the structure of the object examined.

9. The resonances of the system lying within the range of operating rpm of the engine are made apparent by obtaining the vibratory characteristics of the system along with respect to revolutions. Taking into account the location of the "heavy places" along the rotor stages, at these resonance revolutions we determine the location and magnitude (for each stage) of the required balancing weights.

This method was approved in the study of vibrations of full-scale rotors of modern turbo-jet engines on a vacuum vibromeasuring balancing stand. Compressor rotors were balanced the vibration of which exceeded the allowable limits and which is not eliminated by modern factory methods.

As an illustration let us examine the results of the balancing of a rotor by this method. The rotor was balanced on an ordinary balancing stand with respect to the two outer driving planes; the remaining imbalance did not exceed 40 gcm. The engine assembled with this compressor rotor showed increased vibration on the test stand. Attempts to eliminate the vibration by balancing plates, which allowed the compressor rotor to be balanced with respect to the same outer driving planes in the range of operating rpm, did not give the necessary result.

The rotor was examined and rebalanced several times, but each time the engine was tested on the stand, increased vibration was observed, which could not be eliminated by plates.

Before sending the rotor to the MAI, it was again balanced to 40 gcm. The rotor was set up in the vacuum chamber and run up several times to maximum rpm. As a result, the dependence of deflections upon rpm was obtained and the vibratory characteristic of the rotor in the initial state was taken.

After processing the data it was established that the system had two resonance regimes, at 6400 and 7200 rpm (Fig. 11a). In this case the deflection line of the rotor represented the first oscillation shape; maximum deflection occurred at about the middle part of the rotor. Therefore, it was decided to balance the rotor only with respect to the median plane, the third stage. It is important to note that maximum deflection at the middle section was observed at speeds corresponding to two resonances; the directions of the deflections in both cases diverged by about 60° . Two balancing weights were hung alternately. The first weight was hung to eliminate the deflection occurring at 7200 rpm (Fig. 11b) and the second to eliminate the deflection appearing at 6400 rpm (Fig. 11c). As a result, rotor vibration at 7200 rpm was reduced by a factor of eight, and at 6400 rpm, by a factor of two, without changing the vibration level at other speeds.

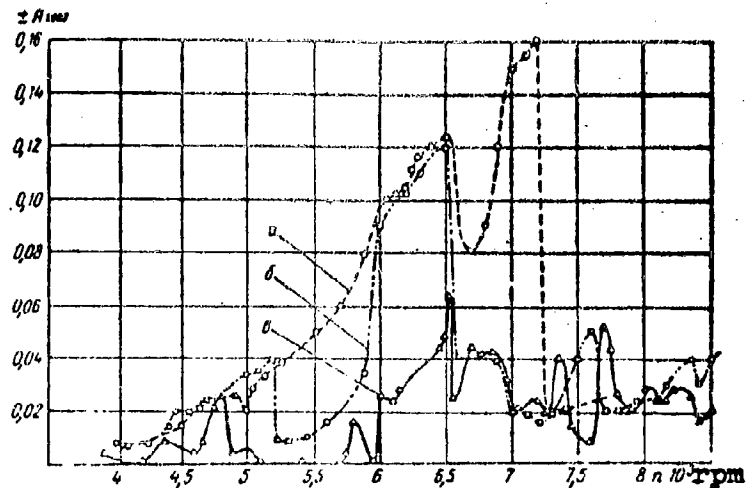


Fig. 11. Change in vibration amplitude with respect to speed at various rotor loads.

a) initial position of rotor; b) with weight $q_1 = 250$ gcm; c) with additional weight $q_2 = 125$ gcm in a new position.

Conclusions

The existing method of balancing rotors by using two outer correction planes can lower the level of vibrations but cannot eliminate their causes and guarantee vibrationless engine operation.

The studies showed that the proposed method of balancing with additional (median) correction planes at operating speeds allows rotor deflection to be decreased considerably, and thus rotor stress and the vibration level are reduced.

On the basis of the studies made, the following order for engine rotor balancing is recommended:

1. The separate static balancing of parts in the rotor assembly.
2. Dynamic balancing at low speed with the two outer driving planes.

3. Final balancing at operating rpm using additional median correction planes.

The last two steps can be carried out on a vibromeasuring balancing stand, which makes it possible to study vibrations in full-scale gas-turbine engines and their assemblies at operating rpm and above with negligible expenditure of power.

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